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# EXPERIMENTAL FLOW ANALYSIS OF AEROSPACE DUCT OF EQUIPMENT COOLING SYSTEM IN AN AIRCRAFT

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Original Article

***ABSTRACT***

***This paper discusses about the Flow analysis of Aerospace ducts of Equipment cooling system of an aircraft. Equipment Cooling is provided to maintain assigned temperature conditions in equipment compartment for satisfactory working of Line Replaceable units (LRUs), which are installed inside equipment compartment. Forced air cooling is provided to these LRUs using Air Conditioning system (ACS) air. The life span of aircraft is approximately 30-40 years, and various up-gradations are required to be carried out to meet the operational &mission requirements. Up-gradation programmes of aircraft required installation of new LRUs, thus a requirement of redistribution of flow through complex duct geometries to meet their additional cooling requirements is mandatory. In the present practice, designers carryout the redistribution of flow of air through these complex ducts using Continuity & Bernoulli's equation. It was experienced that new LRUs at so many times not function properly due to inadequate cooling condition resulting malfunctioning of LRUs. This Paper aims to identify the reason for not achieving the designed cooling condition at designated LRUS. To analyse the flow through Aerospace ducts in transient condition, an experimental Environmental Control System (ECS) test rig has been newly designed to simulate the actual layout of installation of Aerospace ducts into aircraft. Theoretical analysis correlated by experimental trials analysis was carried out on Environmental Control System Test Rig. The results analysed and concluded the reason for not achieving the required cooling to LRUs.***

***KEYWORDS: Equipment Cooling System, Environmental Control System, Flow analysis & Line Replaceable Unit (LRU)***

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## Notations

$\rho$ = Density of air

$\mu$ = Dynamic Viscosity

$\nu$ = Kinematic Viscosity

m* =Mass flow rate

$m_{13}$ = air mass flow rate in branch 13

V= Initial Velocity of the air

f = Darcy friction factor

$R_h$ = Hydraulic radius

$D_h$ = Hydraulic Diameter

*Re* = Reynolds no of flow

$(h_f)_{01}$ = Head loss due To Friction in section 0 to1

$h_f)_{13}$ = Head loss due To Friction in section 1 to3

$V_{01}$ = Mean velocity between section 0 & 1

$V_{13}$ = Mean velocity between section 1 & 3.

$E_0$ = Energy at the section 0

$E_1$ = Energy at the section 1

$E_3$ = Energy at the section 3

$P_0$ = Pressure drop at section 0

$P_1$ = Pressure drop at section 1

$P_3$ = Pressure drop at section 3

$D_{01}$ = Diameter of sections 0 to 1

$D_{13}$ = Diameter of sections 1 to 3

$A_{01}$ = Area of sections 0 to 1

$A_{13}$ = Area of sections 1 to 3

$L_{01}$ = Length of sections 0 to 1

$L_{13}$ = Length of sections 1 to 3

ε = Surface roughness

## 1. INTRODUCTION

This research paper discusses about the Theoretical & Experimental analysis of mass flow rate to maintain assigned temperature conditions in equipment compartment for satisfactory working of Line Replaceable units (LRUs), which are installed inside equipment compartment. Force air cooling is provided to these LRUs using Air Conditioning system (ACS) air. The life span of the aircraft is approximately 30-40 years and during this life spans various up gradation to be carried out to meet the operational & Mission requirements. During up gradation programmes of aircraftdue to introduction of new LRUs, there remains a requirement of redistribution of flow through complex duct geometries to meet their additional cooling requirements. In the present practice, designers carryout the redistribution of flow of air through these complex ducts using Continuity & Bernoulli's equation. But in order to avoid the malfunctioning of the LRUs, it needs to analyse the flow experimentally under transient condition. A new Environmental Control Test Rig is designed and experiment performed. It is also to be noted that to increase or decrease the speed of aircraft, pilots has to change the engine throttle rating. As engine rating changes, the bleed air pressure fed to Air Conditioning System LRUs varies. This

leads to increment or decrement of turbo cooler turbine inlet pressure. Subsequently, based on differential pressure across the turbine of turbo cooler, the mass flow rate varies. The sudden variation of this engine rating leads to variation in flow through ducts and variation of pressure drop frequently due to variable Reynolds's no. Hence, there is a need to study the effects of variation of engine throttle on sudden increment and decrement of flow? Generally, it is very difficult for the designer's to predict the effect of variation of engine throttle on flow through complex duct geometries due to change in transient condition.

## 2. LITERATURE REVIEW

To start a research project, it is very important to know the current methodologies used in industry and explore simple and accurate methodologies for analysis. The first step for flow analysis is through these complex ducts using Continuity & Bernoulli's equation. Initial velocity of air is to be found out for given diameter of pipe by continuity equation. The redistribution of flow is carried out by Bernoulli's equation, for which Reynolds number to be found out. The head loss is important factor for redistribution of flow. Total head loss in Pipe flow is calculated by Darcy-Weisbach formula $\mathbf{h_f = \frac{fLV^2}{2gD}}$, For head loss, friction factor to be calculated. The Darcy friction factor formulae are equations based on experimental data and theory. Darcy-Weisbach friction factor, resistance coefficient or simply —friction factor and is four times larger than the Fanning friction factor [1]. For laminar flow, The Darcy friction factor for laminar flow in a circular pipe (Reynolds number less than 2320) is given by the formula: f $=\frac{64}{Re}$Transition flow(neither fully laminar nor fully turbulent) flow occurs in the range of Reynolds numbers between 2300 and 4000. The value of the Turbulent flow in smooth conduits: The Blasius correlation is the simplest equation for computing the Darcy friction factor. Turbulent flow in rough conduits: The Darcy friction factor for fully turbulent flow (Reynolds number greater than 4000) in rough conduits is given by the Colebrook equation. **Colebrook–White equation**: Colebrook–White equation (or Colebrook equation) expresses the Darcy friction factor(f) as a function of Reynolds number Re and pipe relative roughness ε / $D_h$, fitting the data of experimental studies of turbulent flow in smooth and rough pipes.[2][3] The equation can be used to (iteratively) solve for the Darcy–Weisbach friction factor f. For a conduit flowing completely full of fluid at Reynolds numbers greater than 4000, it is expressed as:

$$\frac{1}{\sqrt{f}} = -2*\log_{10}*\left(\frac{\varepsilon}{3.7Dh} + \frac{2.51}{Re\sqrt{f}}\right)$$

Some sources use a constant of 3.71 in the denominator for the roughness term in the first equation above. [4] Recently, the Lambert W function has been employed to obtain explicit reformulation of the Colebrook equation. [5] Colebrook equation can be solved by iteration using the Newton–Raphson method. Approximation of Cole brook equation is provided by Halland eqaution, Swamee Jain Equation, Serghides's solution and Goudar–Sonnad equation. The Haaland equation was proposed by Norwegian Institute of Technology professor Haaland in 1984. The Haaland equation is defined as: $\frac{1}{\sqrt{f}}= -1.8*\log_{10} [\frac{\varepsilon}{3.7D} + \frac{5.74}{Re^{0.9}})]^{-2.}$

The Swamee–Jain equation [6] is used to solve directly for the Darcy–Weisbach friction factor f for a full-flowing circular pipe. It is an approximation of the implicit Colebrook–White equation. Swamee Jain equation is given by f=0.25 $[\log_{10} ( (\frac{\varepsilon}{3.7D}) +(\frac{5.74}{Re^{0.9}}))]^{-2}$. Goudar equation [8] is the most accurate approximation to solve directly for the Darcy–Weisbach friction factor for a full-flowing circular pipe. It is an approximation of the implicit Colebrook–White equation. Brkić shows one approximation of the Colebrook equation based on the Lambert W-function [9]. Early approximations by Paul

Richard Heinrich Blasius in terms of the Moody friction factor [10] $f = 0.316Re^{0.25}$. Johann Nikuradse in 1932 proposed that this corresponds to a power law correlation for the fluid velocity profile. Mishra and Gupta in 1979 proposed a correction for curved or helically coiled tubes, taking into account the equivalent curve radius. From literature survey, it is concluded that Swamee Jain equation is latest refined available equation considering all the relevant factors. Hence, Swamee Jain equation was selected for our application.

### 2.1 Equipment Cooling System during up Gradation Programme

During up gradation of aircraft, old avionics LRUs have been replaced by the new one like: UNIT 1, UNIT 2, UNIT 3, UNIT 4, UNIT 5, UNIT 6, UNIT 7 & UNIT 8. The cooling air requirement of these LRUs along with their positioning on aircraft is placed at **Table 1**.

**Table 1: Cooling Air Requirement of these LRUs along with their Positioning on Aircraft**

| LRUs Name (Newly Introduced) | Required Air Mass Flow Rate Through LRU (Kg/hr) |
|---|---|
| UNIT 1 | 20 |
| UNIT 2 | 20 |
| UNIT 3 | 55 |
| UNIT 4 | 20 |
| UNIT 5 | 65 |
| UNIT 6 | 164 |
| UNIT 7 | 14 |
| UNIT 8 | 15 |

Configuration of placement of New LRUs at Nose Compartment of aircraft is shown at **Figure 1.**

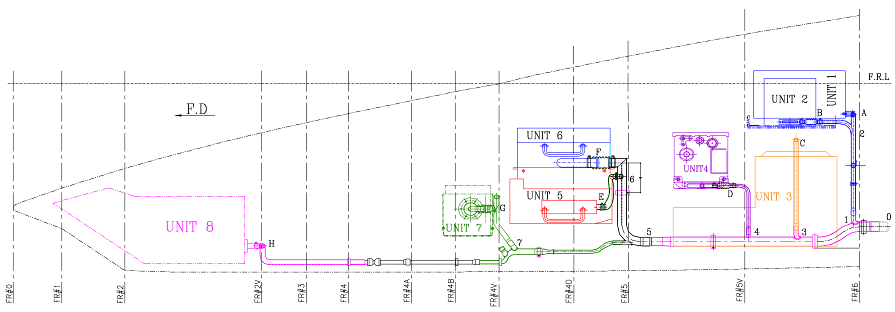


**Figure 1: Configuration of Equipment Cooling System**

Detailed flow analysis regarding distribution of air through various LRUs of aircraft is carried out with the help of Continuity & Bernoulli's equation, to ensure appropriate flow distribution in respective LRUs. Head losses due to friction & bends during flow were calculated using friction factor $f = 0.0032 + \frac{0.210}{Re^{0.257}}$.

A sample theoretical calculation based on available mass flow rate is shown below:

Air mass Flow Rate outlet of turbo cooler = **400kg/hr**

Mass flow rate of cooling air supplied to Compartment 5 = 18 kg/hr

Mass flow rate of cooling air supplied to newly introduced LRUs = 400-18 = 382 Kg/hr

Absolute outlet Pressure from Turbo cooler = **1.2 kgf/cm²**

Temperature of the air outlet from turbo cooler ≈**$15^0$ C**

Properties of air at initial temperature are taken from thermodynamic table.

Density of air **$\rho$ = 1.2kg/m³**

Dynamic Viscosity $\mu$ = **$1.81 \times 10^{-5}$ N-s/m²**

Kinematic Viscosity **$\nu$ = $1.51 \times 10^{-5}$ m²/sec**

Mass flow rate **m* = ρ*A*V (Applying Continuity Equation)**

$382/3600 = 1.2*\frac{\Pi}{4}*D^2*V$

Taking the initial Diameter of the pipe as 36mm= 0.036m

Initial Velocity of the air **V = 86.91 m/s**

Assumed that there is no pressure drop through pipelines during the flow due to less pressure gradient across turbo cooler outlet & newly introduced LRUs. The flow is assumed to be due to dynamic pressure gradient (i. e kinetic head gradient).

To find the head loss across the pipe lines, we will calculate the friction Factor **'f'**

To calculate f, it is required calculate Reynolds no of flow:

$Re = \rho*D*V/\mu$

**Re** = $1.2*0.036*86.91/1.81*10^{-5}$

**Re** = 207431.6

As Re> 4000, the flow is Turbulent.

For 105< **Re** < **$4 \times 10^7$** using the empirical relation for friction factor f

**$f = 0.0032 + 0.210/ Re^{0.257}$**

$f = 0.0032 + 0.210/ (207431.6)^{0.257}$

***f = 0.01223***

## 3. CALCULATIONS FOR SECTION 0 TO 1


Energy balance equation between section 0 and 1

**$E_0 = E_1 + (h_f)_{01}$**

Using Bernoulli's equation between section 0 & 1

$\frac{P_0}{\rho g} + \frac{V_0^2}{2g} + Z_0 = \frac{P_1}{\rho g} + \frac{V_1^2}{2g} + Z_1$ + **head Losses**

Due to less pressure gradient across turbo cooler outlet & New LRUs, itis assumed that pressure drop between sections 0 to 1 is negligible. Thus,

$\frac{P_0}{\rho g} = \frac{P_1}{\rho g}$, & $Z_0 = Z_1$

$L_{01}$ = 0.085 m, $D_{01}$ = 0.036m, $V_{01}$ =86.91 m/s

**$V_{01}$:** mean velocity between section 0 & 1.

$\frac{V_0^2}{2g}$- **head Losses**= $\frac{V_1^2}{2g}$=**$E_1$**

Head loss due To Friction in section 0 to1,

$(h_f)_{01} = f * L_{01} * V^2_{01} / 2gD_{01\ n}$

$(h_f)_{01}$ =0.01223*0.085*86.91$^2$ / 2*9.81*0.36

$(h_f)_{01}$ = 1.112 m of air

Velocity Head Available at the inlet = $V^2_{01}$ /2g= 86.91$^2$ / 2 * 9.81= **384.98**

Hence, Energy at the section 1

**$E_1 = E_0 - (h_f)_{01}$**

$E_1$ = 384.98 –1.112

**$E_1$ = 383.868 m of Air**

In a similar manner, calculations for remaining sections have been undertaken and results are tabulated in **Table 2**.

**Table 2: Calculated Diameters of Pipelines at Respective Section Supplying ECS Air to LRUs**

| Section | Mass Flow Rate Kg/hr | Length of Pipe (m) | Energy at the Inlet of Section ($E_{inlet}$) m of air | Frictional Loss m of air | Energy at Outlet of Section ($E_1$) = $E_{inlet}$- FL | Iterative Velocity V (m/s) | Calculated Diameter (mm) | Fabricated Pipe (mm) |
|---|---|---|---|---|---|---|---|---|
| 0-1 | 382 | 0.085 | 385.4 | 1.112 | 383 | 86.91 | 35.9 | 36 |
| 1 to 3 | 332 | 0.305 | 383 | 37.98 | 345.01 | 82.2 | 34.5 | 36 |
| 1 to 2 | 50 | 0.385 | 383 | 97.59 | 285.4 | 74.8 | 14.03 | 14 |
| 2 to A | 25 | 0.23 | 285.40 | 60.98 | 224.42 | 66.3 | 10.54 | 12 |
| 2 to B | 25 | 0.062 | 285.40 | 20.27 | 265.13 | 72.1 | 10.10 | 12 |
| 3 to 4 | 291.8 | 0.175 | 345.01 | 20.60 | 324.40 | 79.7 | 34.5 | 34 |
| 3 to C | 40.2 | 0.41 | 345.01 | 76.42 | 268.58 | 58.8 | 14.19 | 16 |
| 4 to 5 | 266.8 | 0.46 | 324.4 | 47.12 | 277.28 | 72.6 | 32.9 | 32 |
| 4 to D | 25 | 0.375 | 324.4 | 67.30 | 257.09 | 56.6 | 11.41 | 10 |
| 5 to 6 | 232 | 0.14 | 277.3 | 14.33 | 262.94 | 70.6 | 31.12 | 32 |
| 6 to E | 55 | 0.195 | 262.9 | 21.69 | 241.25 | 54.71 | 17.21 | 18 |
| 6 to F | 177 | 0.4 | 262.9 | 24.42 | 238.52 | 54.4 | 30.97 | 32 |
| 5 to 7 | 34.8 | 0.46 | 277.2 | 81.92 | 195.35 | 60.3 | 13.04 | 12 |
| 7 to G | 17 | 0.32 | 195.3 | 36.50 | 158.85 | 43.7 | 10.7 | 12 |
| 7 to H | 17 | 1.045 | 195.3 | 99.19 | 96.16 | 40.6 | 11.1 | 12 |

### 3.1 Experimental Determination of Velocity of air flow at Discrete Air Outlet Points of Air Conditioning System and Mass Flow Rate of Cooled Air Supplied to Avionics LRUs

Velocity of Air flow at discrete air outlet points of Air Conditioning System i. e. Inlet to LRUs has been experimentally determined by designing an Environmental Control System (ECS) test rig as described below:

#### 3.1.1 Experimental Setup of ECS Test Rig

Aerospace ducts based on theoretical calculations mentioned at **Table 2** have been manufactured. These ducts were made of Aluminum alloy. An experimental Environmental Control System (ECS) test rig has been designed to simulate the actual layout of installation of Aerospace ducts into aircraft as shown in **Figure 2.**

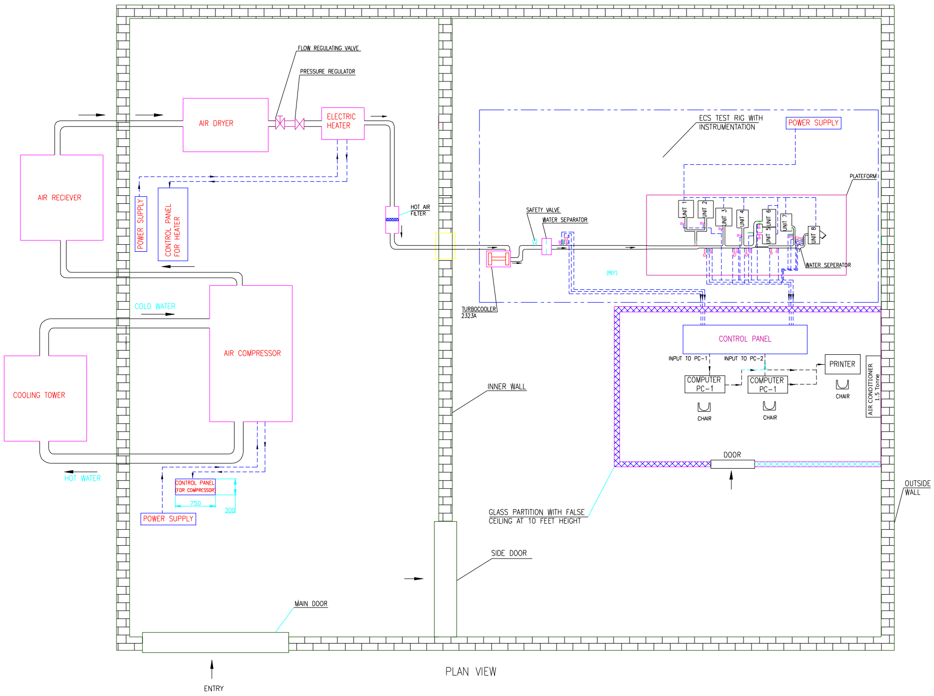


**Figure 2: ECS Test Rig Layout**

Aircraft turbo cooler is used on ECS test rig to simulate the system parameters based on variable input pressure and temperature. Thermal flow meters were installed at the duct to measure the Air flow rate supplied to particular LRU. To record and online recording of test rig parameters, PLC based control system has been designed. Physical components of ECS test rig are shown in **Figure 3**. Placement of thermal flow meters at the inlet of respective LRUs is shown in **Figure 4.**


**Figure 3: Components of ECS Test Rig**

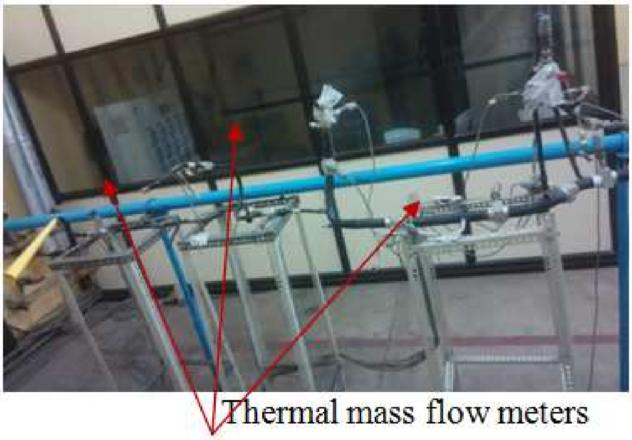


**Figure 4: Installation of Flow Meter at the Inlet of ECS ducts**

### 3.2 Experimentally Determination of Mass Flow Rate of Cooled Air Supplied to Avionics LRUs

Mass flow rates supplied to particular LRUs have been measured using thermal flow meters on ECS test rig shown at Figure 2. The results are placed in **Table 3**

**Table 3: Experimentally Measured Mass Flow Rates Supplied to Respective LRUs**

| LRUs Name | Required Air Mass Flow Rate Through LRU (Kg/hr) | Theoretical Calculated Mass Flow Rate Through LRU(Kg/hr) | Experimentally Measured Mass Flow Rate(Kg/hr) |
|---|---|---|---|
| UNIT 1 | 20 | 21 | 32 |
| UNIT 2 | 20 | 21 | 28 |
| UNIT 3 | 55 | 57 | 43 |
| UNIT 4 | 20 | 25 | 31 |
| UNIT 5 | 65 | 66 | 55.5 |
| UNIT 6 | 164 | 166 | 180 |
| UNIT 7 | 14 | 17 | 16.5 |
| UNIT 8 | 15 | 17 | 16 |

**Table 4: The Comparative Deviation of Mass Flow Rates Based on Theoretical**

| Section | Deviation, % | | |
|---|---|---|---|
| | (Theoretical Mass Flow Rate) Kg/hr | (Experimental Mass Flow Rate) Kg/hr) Average Values Taken after 05 Times Experiment Run | Theoretical w. r. t Experimental Mass Flow Rate |
| 0-1 | 382 | 381.5 | 0.131062 |
| 1 TO 3 | 332 | 334 | -0.5988 |
| 1 TO 2 | 50 | 44 | 13.63636 |
| 2 TO A | 25 | 32 | -21.875 |
| 2 TO B | 25 | 28 | -10.7143 |
| 3 TO 4 | 291.8 | 296 | -1.41892 |
| 3 TO C | 40.2 | 43 | -6.51163 |
| 4 TO 5 | 266.8 | 261 | 2.222222 |
| 4TO D | 25 | 31 | -19.3548 |
| 5 TO 6 | 232 | 219 | 5.936073 |
| 6 TO E | 55 | 55.5 | -0.9009 |
| 6 TO F | 177 | 180 | -1.66667 |
| 5 TO 7 | 34.8 | 32 | 8.75 |
| 7 TO G | 17 | 16 | 6.25 |
| 7 TO H | 17 | 16 | 6.25 |

## 4. RESULTS AND DISCUSSIONS

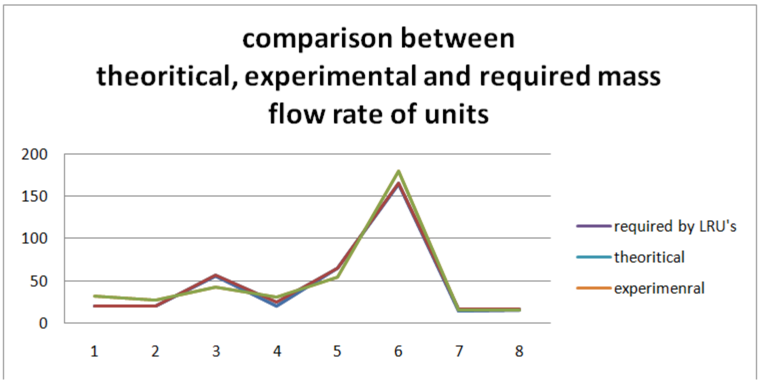


**Figure 5: Diameter Vs Deviation in Mass Flow Rate**

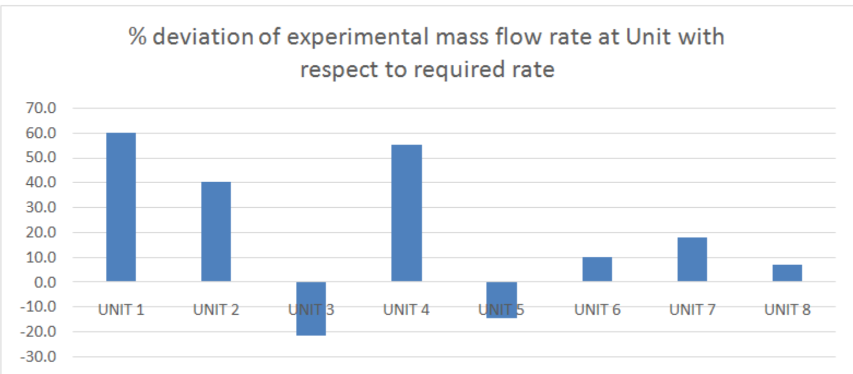


**Figure 6: % Deviation of Experimental Mass Flow Rate at Unit with Respect to Required Rate**

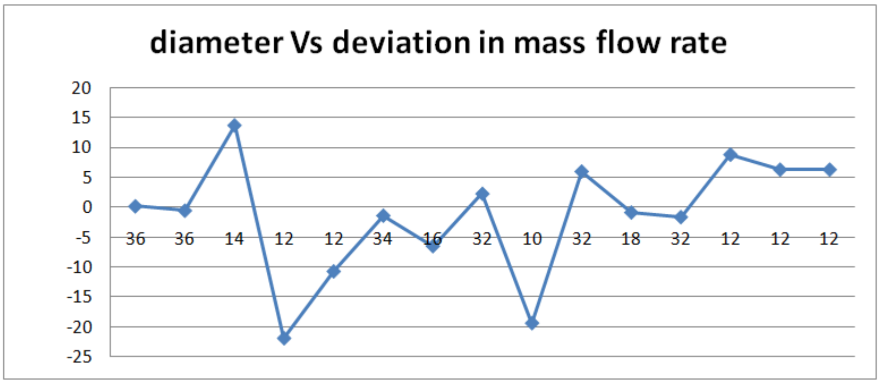


**Figure 7: Diameter Vs Deviation in Mass Flow Rate**

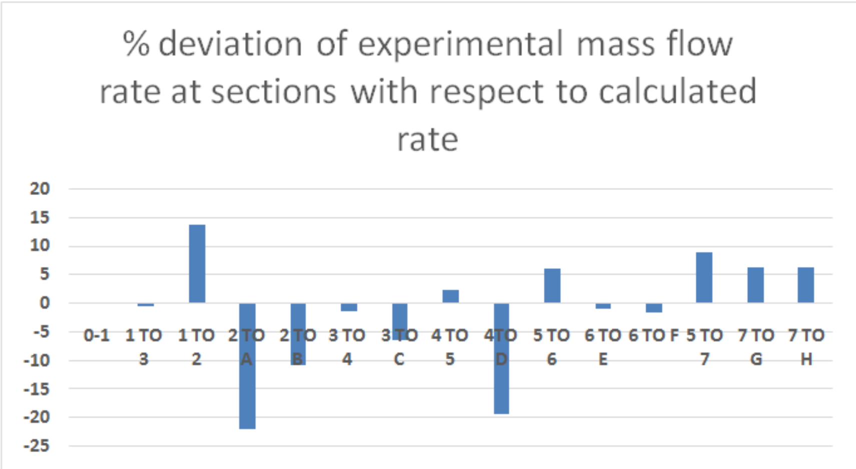


**Figure 8: % Deviation of Experimental Mass Flow Rate at Sections with Respect to Calculated Rate**

Engineering practices for pipe systems required calculation of mass flow rate and head loss. In this experimental setup, we have calculated mass flow rate to each LRUs and complex duct geometry. The experimental mass flow rate is compared with the theoretical and required one. It is observed that the mass flow rate is more in case of unit 1, 2, 4, and 6. In case of unit 3 and 5, the amount of mass flow rate is less than the required one.

Figure 6 represents, the graph between % deviation of experimental mass flow rate at Unit with respect to required mass flow rate and observed significant variations from -21% to 60%. This % deviation in mass flow rate in various units is due to the abrupt change in diameter of ducts and the position of the unit in the complex duct geometry. Figure 8 represents the graph between % deviation of experimental mass flow rate at sections with respect to calculated rate and observed that the deviation are due to the abrupt change in diameter of duct. The graph between diameters of the complex duct geometry and percentage deviation in mass flow rate has plotted and observed that the percentage deviation is more in case of abrupt changes in pipe diameter from large diameter pipes to the smaller diameter.

## 5. CONCLUSIONS

This paper presents the experimental results of mass flow rate to the LRU's and the complex duct geometry of aircraft. From the results, it has been concluded that there is variation in mass flow rate required by the LRU's as suggested by original unit manufacture. The variation in mass flow rate in complex duct geometries is observed, which leads to

improper distribution of flow to the LRU's, cause the malfunctioning of LRU's. For the safety and functioning of LRU's, the assumptions taken by the designers for redistribution of flow does not hold good. This again needs to be analyzed.

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